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DESIGN AND PERFORMANCE EVALUATION OF A CRYOGENIC CONDENSER FOR AN IN-PILE EXPERIMENT

by Robert W. Graham, Richard J. Crum and Yih-Yun Hsu

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

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SUMMARY

An apparatus was designed to enable in-pile irradiation of materials in liquid hydrogen at cryogenic temperatures. One of the principal components of this apparatus was a horizontal tube condenser. The performance of the condenser was evaluated by running a liquid-nitrogen prototype of the apparatus at heat loads comparable to or greater than those expected during the irradiation. The tests showed that the condenser was capable of handling the design heat load and that the design procedure was sound.

INTRODUCTION

As the result of a proposal to do some material studies in pile in liquid hydrogen, the design of a hermetically sealed test chamber suitable for insertion in the Plum Brook Reactor was inaugurated. One of the design criteria to maximize safety of operation was that the hydrogen coolant be confined in a welded, lead-free test chamber and recycled; consequently, a condenser was selected to be a major constituent of the apparatus.

A survey of the design literature revealed that there was practically no information relative to hydrogen condensing. What little was found pertained to vertical condenser tubes and the design of concern would have to involve horizontal tubes.

Of course, there is considerable literature pertaining to condensers in general - primarily those involving steam. Condensing-heat-transfer correlations do exist, but their applicability to an entirely different thermal domain must be questioned. As a means of evaluating the design procedure for hydrogen, it was considered advisable to design and test a nitrogen condenser prototype similar to the hydrogen one. Running this nitrogen prototype condenser out of pile would yield valuable information about the applicability of the correlations and scaling laws employed. This report contains an evaluation of the nitrogen condenser performance with reference to its design. These

tests with the nitrogen condenser indicate that the design equations and procedure can be applied to the hydrogen condenser with considerable confidence and the desired performance of the condenser will be achieved.

APPARATUS AND PROCEDURE

Figure 1 shows the design concept of the apparatus as it was envisioned for the actual in-pile experiments where liquid hydrogen was the coolant. The material specimens to be tested would be placed in the liquid-hydrogen sump shown at the left side of the drawing. The nominal liquid-hydrogen temperature in the sump is 30 K (54° R). The condenser part of the apparatus comprises 15 horizontal tubes in which helium flows as a coolant. The helium is to enter the condenser tubes at 24 K (43° R). The anticipated heat load to be dissipated by the condenser is approximately 845 watts (2880 Btu/hr). This heat load is the estimated gamma and neutron heating that will be developed in the test hole of the reactor.

In carrying out the nitrogen prototype test program, electrical heaters were installed in the heat exchanger to simulate the gamma and neutron heating that would be experienced in pile. Sufficient electrical heating capacity was installed in the nitrogen heat exchanger to far exceed the anticipated design level. Since liquid nitrogen was to be used as the coolant in the prototype heat exchanger instead of gaseous helium, the design-point heat flux for the prototype should be higher than that of the actual in-pile heat exchanger. The higher density and Prandtl number of the liquid nitrogen would effect a greater transport of heat for comparable temperature differences and flow velocities. A number of ways of establishing this prototype heat load could be used. Through simulitude relations, one could compute the heat load on the condensing side and on the coolant side for various conditions. It was decided that the most conservative estimation would be to operate the nitrogen prototype at the same value of the flooding parameter φ (defined in eq. (3)) as would be realized with the hydrogen-helium heat exchanger. It turned out that the design heat load would be approximately 2400 watts (8200 Btu/hr) for the prototype. As is shown in table I, the maximum test heat load was 6000 watts (20 500 Btu/hr) or about $2\frac{1}{2}$ times the design value. The heat capacity was purposely pushed to this high value to assess the margin of performance of the heat exchanger. It was felt that if the prototype would operate satisfactorily beyond its design point, the actual in-pile heat-exchanger design would be satisfactory.

Obviously, in the nitrogen prototype tests it was impossible to simulate very closely all the conditions that would occur in the hydrogen apparatus. Such things as fluid and wall temperatures, liquid flow rates, and heat-transfer coefficients would be different. For example, at the coolant tube entrance, typical computed conditions for the prototype and the hydrogen heat exchanger are as follows:

Exchanger	Coolant temperature, K	Wall temperature, K	Estimated coolant-side heat-transfer coefficient, $W/(m^2)(K)$
Prototype	80	94	3060
Hydrogen	23.8	24.8	1300

Figure 2 shows the apparatus and how it was instrumented for the nitrogen experimental program. A liquid-level indicator was installed in the sump to give an approximate idea of the amount of liquid nitrogen there. The instrument consisted of six carbon resistor circuits located at different positions on the probe. The sump was utilized as a convenient reference temperature bath for the cold junctions of the thermocouple instrumentation.

The thermal conditions of the condenser coolant (liquid nitrogen) were determined from platinum resistor bulk temperature measurements as the coolant entered the condenser and when it discharged. A turbine-type flowmeter was used to monitor the flow rate. The liquid nitrogen pressures at the inlet and exit were measured.

As is shown in figure 2, one of the condenser tubes was instrumented with several types of thermocouple sensors. The temperature difference ΔT between the outside wall and the vapor surrounding the tube was measured at three axial stations, 14.7, 36.8, and 61.5 centimeters (5.8, 14.5, and 23.2 in.) from the inlet. Centerline ΔT thermocouples were installed at four internal stations of the condenser tube. As shown in figure 2 they were located at 14.7, 29.4, 44.2, and 58.8 centimeters (5.8, 11.6, 17.4, and 23.2 in.) from the inlet.

During the course of the test program, some of these thermocouples shorted or opened up and could not be repaired. Thus, not all of the instrument stations of the condenser tube were available during the testing. However, a sufficient number remained intact to yield performance information.

HEAT-TRANSFER DESIGN CALCULATIONS

The correlations and important assumptions pertinent to the design of the hydrogen apparatus and the nitrogen prototype are outlined in this section. Most of these correlations, when taken from their original references, were designated for use with U. S. customary units. They will be cited in their original published form. In cases where the predictions are dimensional, the conversion factors necessary to list them in SI units are presented.

Heat-Transfer Coefficient Inside Condenser Tubes

Hydrogen apparatus. - In the hydrogen apparatus, cold helium gas was to be used as the condenser coolant. A simple, forced-convection correlation for gases obtained from reference 1 was employed in estimating the inside-tube heat-transfer coefficients:

$$h = \kappa G^{0.8} d^{-0.2} \sqrt{\frac{T_b}{T_w}} \tag{1}$$

where

h heat-transfer coefficient, $Btu/(ft^2)(hr)(^oF)$; to convert to $W/(m^2)(K)$ multiply by 5.69

κ proportionality constant which includes fluid properties as well as Dittus-Boelter constant, $C\left(k^{0.6} c_p^{0.4}/\mu^{0.4}\right)$; $\kappa = 0.020$ for helium

G flow per unit area

d diameter or hydraulic diameter

Nitrogen prototype. - Liquid nitrogen was employed as the condenser coolant. A conventional forced-convection correlation for liquids was utilized, namely,

$$h = \frac{k}{d} C Re^{0.8} Pr^{0.25} \tag{1a}$$

where Re is the Reynolds number, Pr is the Prandtl number, and C is a coefficient.

Condensing Heat-Transfer Coefficient Outside Tubes

Jakob (ref. 2, p. 673) suggests the following correlation for condensation with multiple horizontal tubes:

$$h = 0.725 \sqrt[4]{\frac{\rho^2 \lambda k^3 g}{n\mu d(t_f - t_s)}} \tag{2}$$

where

h heat-transfer coefficient, $Btu/(hr)(ft^2)(^oF)$; convert to $W/(m^2)(K)$ by multiplying by 5.69

ρ density, lbm/ft^3

λ heat of vaporization, Btu/lb

k thermal conductivity, Btu/(ft)(hr)(^{o}F)

g gravity, ft/sec^2

n number of tubes

μ viscosity, lb/(ft)(hr)

d tube diameter, ft

t_f saturation temperature, ^{o}F

t_s surface temperature, ^{o}F

Flooding or Holdup Estimation

Perhaps the most critical and questionable estimation in the entire design procedure is the one pertaining to the prediction of possible "flooding" in the condenser. By "flooding" we mean that the condensate is inhibited from returning to the sump by the moving vapor. A flooding criterion expressly developed for vertical condenser tubes was used (ref. 3). No criterion for horizontal tubes was found in the literature. The flooding criterion is

$$j_g^{*1/2} + j_f^{*1/2} = \varphi = 0.77 \tag{3}$$

in which

$$j_g^* = j_g \rho_g^{1/2} \left[g d_s(\rho_f - \rho_g)\right]^{-1/2}$$

$$j_f^* = j_f \rho_f^{1/2} \left[g d_s(\rho_f - \rho_g)\right]^{-1/2}$$

where

j_g upward gas flow, ft/sec

ρ_g density of vapor

g local gravity

d_s horizontal spacing between tubes

ρ_f density of liquid

j_f downward liquid flow, ft/sec

In using this criterion, we assumed that at the holdup condition, no liquid condensate was dropping, or $j_f^* = 0$.

RESULTS AND DISCUSSION

Twenty-four runs were made with the nitrogen prototype heat exchanger. The electrical heat input to simulate the gamma and neutron heating ranged from 500 to 6000 watts. The design heat load for the nitrogen prototype was approximately 2400 watts.

A computer code was developed incorporating the correlations discussed in the last section. The output of the code was the heat extracted in the condensing process. Table I summarizes the principal operating conditions, including the calculated and measured electrical heat to the apparatus. Generally, when the calculated and experimental heat inputs are compared, they are in relatively good agreement. One can conclude that the design analysis is fairly accurate as long as steady-state operation takes place. There remains the question of flooding or holdup, which would make the condenser inoperative.

It was estimated by the analysis of reference 3 that the threshold of flooding would not be encountered until heat rates $2\frac{1}{2}$ times the design value were realized. The unusual geometry of the heat exchanger quadrant did raise some doubts whether the flooding criterion was meaningful. It was suggested that a kind of horizontal flow holdup could occur. This speculation meant that the condensate accumulated in the bottom of the smaller diameter shell (fig. 1, right side of drawing) could be backed up by a counterflow of the vapor. This situation could result in a flooding excursion and an inoperative condenser. To test this idea, a clear plastic model of the apparatus was built, and the vapor flow was simulated by a series of air jets and the liquid flow by water jets. Over a wide range of conditions (gas and liquid flow rates) there was no evidence for the beginning of a holdup transient. The volumetric rates of the gas and liquid were comparable to the actual ones in the heat exchanger. From this test and from the analysis done on estimating holdup, it is concluded that holdup is not a likely possibility.

As was mentioned in the section APPARATUS AND PROCEDURE, some difficulty was incurred with several of the thermocouples in the instrumented condenser tube. Only two of the internal centerline thermocouples remained operable throughout the entire testing program. However, from knowledge of the inlet and exit coolant temperatures it was possible to gain valuable information about the bulk temperature distribution along the length of the instrumented tube.

Figure 3 shows these bulk temperature distributions for ranges of flow rates and heat fluxes. It is evident that the distributions are not linear, especially near the rear of the heat exchanger where the sump is located. However, if one did assume a linear distribution of bulk temperatures between the inlet and exit stations, it would not be a

bad approximation and would yield fairly good overall results. Such an assumption would not reveal that the rear section of the condenser above the sump is doing most of the condensing, as would be anticipated from an examination of the geometry.

CONCLUSION

The main conclusion from a design and performance evaluation is that a heat exchanger with a horizontal condenser bank can be designed by using standard heat-transfer correlations and that the condenser will operate stably without holdup over a considerable heating range. The condenser described in this report handled 4.6 watts per square centimeter (30 W/in.^{2}), which was approximately $2\frac{1}{2}$ times its design capacity.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, April 7, 1972,
111-05.

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3. Wallis, Graham: One-Dimensional Two-Phase Flow. McGraw-Hill Book Co., Inc., 1969.

TABLE I. - SUMMARY OF NITROGEN-HEAT-EXCHANGER PERFORMANCE

Flow		Chamber pressure		Chamber saturation temperature		Heat, Q, W		$\frac{Q_{calculated}}{Q_{actual}} \times 100$, percent
m^3/hr	gal/min	N/cm^2	$lb/in.^2$ abs	K	^oR	Actual	Calculated	
17.0	74.4	12.05	17.48	78.97	142.05	500	445	89
18.2	79.8	12.36	17.93	79.20	142.45	1000	940	94
17.0	74.4	14.09	20.44	80.39	144.60	2000	1825	91
14.7	64.4	18.29	26.53	82.89	149.10	4000	3528	88
12.3	54.0	21.77	31.58	84.68	152.30	4000	3612	90
2.5	11.0	29.94	43.43	88.01	158.30	2500	2863	115
2.4	10.4	59.14	85.78	97.96	176.20	5000	5607	112
5.0	21.9	45.14	65.48	92.68	166.70	6000	6592	110
2.8	12.5	29.27	42.46	87.79	157.90	2500	2933	117
2.7	12.0	55.44	80.41	95.24	171.30	5500	5228	95
4.9	21.7	49.23	71.41	93.74	168.60	6000	6771	113
2.7	11.6	28.47	41.30	87.48	157.35	2500	2747	110
2.5	10.9	43.50	63.10	92.29	166.00	4000	4068	102
2.4	10.6	55.64	80.70	95.46	171.70	5000	4954	99
2.3	10.2	72.39	105.00	99.02	178.10	6000	5791	97
1.1	5.0	42.68	61.90	92.01	165.50	2500	2340	94
1.0	4.5	54.05	78.40	95.07	171.00	3000	2607	87
3.5	15.4	24.33	35.30	85.79	154.30	2500	2720	109
3.5	15.2	44.81	65.00	92.68	166.70	5000	5238	105
3.4	14.8	58.67	85.10	96.07	172.80	6000	6358	106
2.7	11.9	27.64	40.10	87.18	156.80	2500	2733	109

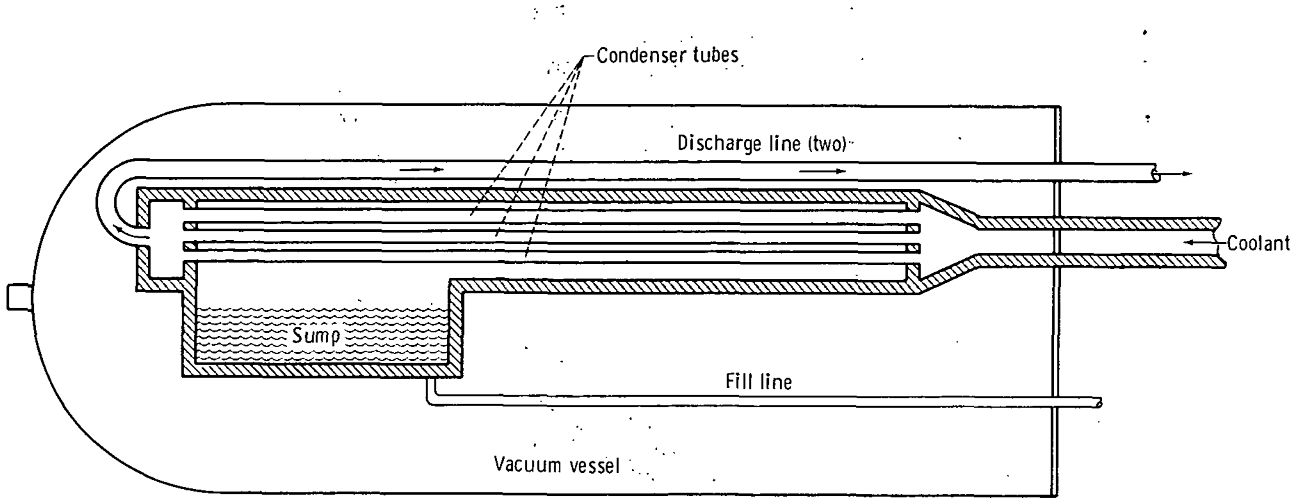


Figure 1. - Schematic of apparatus.

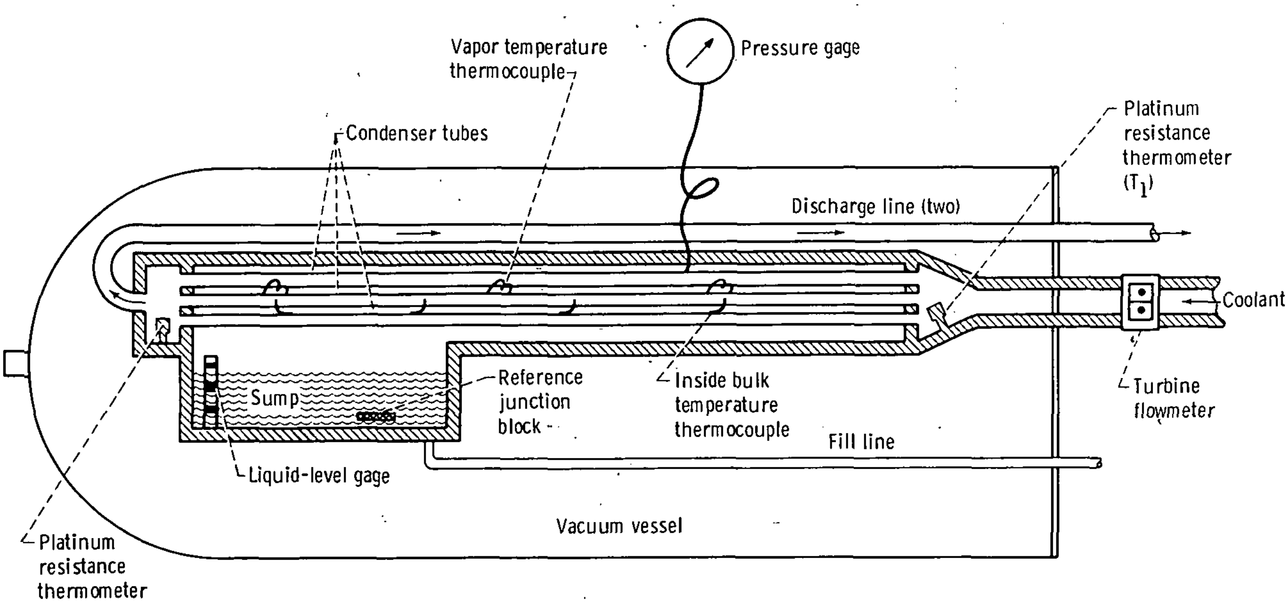
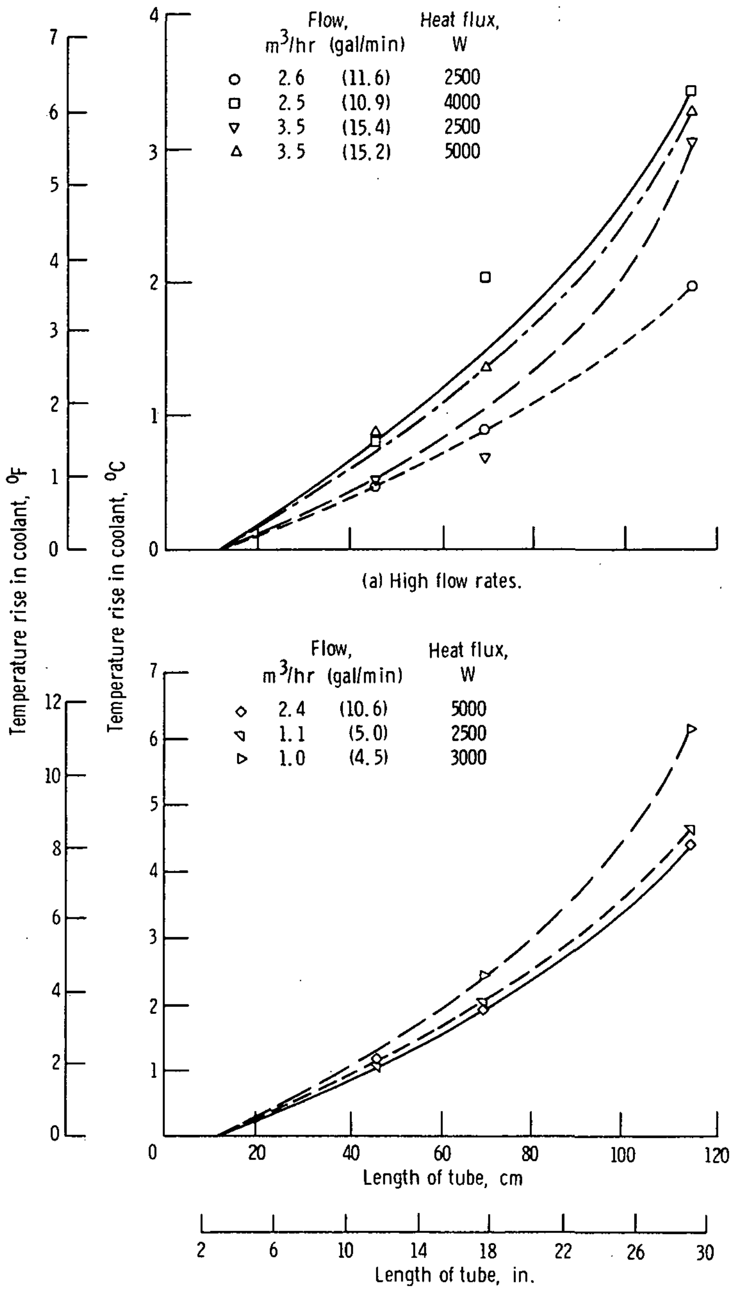


Figure 2. - Instrumentation of condenser tube.



(b) Low flow rates.

Figure 3. - Nitrogen coolant temperature rise tube-in-shell cryogenic condenser.

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